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HALL-SCOTT



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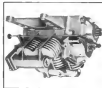


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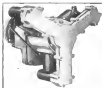
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APRIL 1, 1917

AVIATION AND AERONAUTICAL ENGINEERING

VOL. II. NO. 5

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Editorial Assistant
WILLIAM C. LINDSEY, S.M.
Managing Editor
ROBERT M. WILLIAMS, S.E.

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No. 5

THIS article on "America's Aeronautic Needs and Possibilities" by Dr. Charles D. Wakett which appears in this issue of AVIATION, the AERONAUTICAL ENGINEER, should go far to clear up the confusion which has existed in the industry as well as in lay minds regarding the problems which must be solved if the United States is to build up a real aerial arm. The article shows, best of all, the neglect from which the industry has suffered in the past. The Army and Navy together ordered less than 100 machines in the eight years prior to 1918.

At the outbreak of the European war the British air service was one of the Americas today. The Royal Flying Corps had only eighty six serviceable machines, and of these only six were sent to France.

Excerpted from THE ENGINEER, W. H.

The air service on the continent proved itself the machine, little or without which the usefulness of the infantry and field artillery, at least was greatly reduced if not practically nullified. So Great Britain set about to get airplanes.

In its efforts to get airplanes Great Britain made almost every conceivable mistake. Progress and improvement were discouraged by the Royal Aircraft Factory's authorities. The Royal Flying Corps and the Royal Naval Air Service did not cooperate and in fact competed with one another in a manner which made progress slow but did not produce machines. The respective departments of the two services failed to cooperate and as a result all sorts of different standards and tests were laid down. There was much such as duplication of effort and waste of time. These mistakes are appreciated in Washington today. Great Britain's errors are serving to point out the straight road to the United States.

Great Britain has solved her biggest problem. Today airplanes are being delivered in large quantities to the R. F. C., and the R. N. A. S. This result is largely due to the fact that not practically every order for planes requires a part of a deposit at twenty-five per cent in cash was made by the government with the manufacturer.

This practice is specifically forbidden by law at the present time in this country. A part of Section 3648 of the Revised Statutes, United States, reads:

"No advance of public money shall be made by any one machine. And no sum of money for the performance of any service or the delivery of articles of any description for the use of the United States, private shall not exceed the value of the service rendered, or of the articles delivered, previous to such payment."

However, there is no question that Congress by special legislation can authorize advance of money. This policy has been tried by Great Britain and it has proved successful. It is a policy which would help to solve the greatest problem now confronting the United States Army and Navy air services, which is the problem of how to get better machines, especially machines of military type.

THE PRIMARILY PROBLEM

The paramount question, the problem of getting enough there, depends first of all on the solution of the problem of how to get machines. Secondly, comes the question of getting flying instructors, which presents considerable difficulties. The art of teaching others to fly is a different art from the mere flying of a machine. Thirdly, the question of getting airplane mechanics arises. Good airplane mechanics are rare in America and it is a mistaken notion that a knowledge of an automobile engine fits a man to over for and repair airplane engines. The latter are far more delicate and they require quite different handling. An airplane mechanic, too, should have a thorough knowledge of how to set up and align an airplane. He should understand the installation and care of all bearing, drift and control wires.

Lastly, comes the question of getting trained military pilots. According to Dr. Wakett's article the training of a pilot takes about nine months, involves the wearing out of an airplane and takes one-fifth of the time of a flying instructor. After 1919 Dr. Wakett's program cost applies the instruction of 3,000 aviators yearly. To accept Dr. Wakett's figure of machine needs that 2,000 airplanes will have to be built yearly for training purposes as well as fighting machines of every type by an industry whose total production for 1937 is likely estimated to be 300 machines for the American Army and Navy. Such figures give a bare indication of the future in store for the industry in the military field alone.

To sum up the situation briefly today, the greatest need of the United States is for more facilities to produce airplanes. Until a sufficient number of airplanes can be produced the patriotic offers of the thousands of young men who wish to learn to fly can not be accepted. Only a limited number can be trained because there are in this country today only a few firms competent to give instruction, because this country has only a comparatively small number of trained airplane mechanics and because the output of airplanes is limited by the inadequate manufacturing facilities.

Abstract by E. Z. Gorbach, U. S. N.

It is the custom for manufacturers to taper their struts, but this is unduly done by eye, and the tapered struts in use to-day are unlikely to be very efficient struts where these defects have been associated by unfortunate failures. The following method proposed to determine by a graphical process the correct taper for a strut such that, when loaded as a column, it shall have uniform strength throughout its length.

The graphical process by which the proper sections are determined for a strut of uniform strength is carried out by means of the integrator, and is based on the two following formulae:

$$-E I \frac{d^2 y}{dx^2} = W y \quad (1)$$

$$f = -E I \frac{d^2 y}{dx^2} \quad (2)$$

where

E = modulus of elasticity of the material

I = the (least) moment of inertia at any section distance x from the origin

W = the critical load for the strut

y = deflection at any section

for a strut loaded as in the diagram, (Fig. 1) and

f = the maximum stress at any section

k = the distance from origin axis to most strained fiber

and $\frac{d^2 y}{dx^2}$ = the second derivative of the elastic curve of the strut.

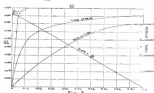
Then with the strut of length $= 2L$ and symmetrical about the line $x = L$, the following conditions hold:

$$\text{when } x = 0, \frac{dy}{dx} = 0$$

$$\frac{dy}{dx} = \text{some initial value} = k$$

$$y = 0$$

$$\text{and when } x = L, \frac{dy}{dx} = 0$$



If we assume values of W and have a table or curve showing the values of I at the various sections, we may draw by a process of trial and error, curves that will satisfy the conditions. The correct value of W or the crippling load for the strut will be that one which causes the $\frac{d^2 y}{dx^2}$ curve to meet the

axis at mid-length of the strut. But we are desirous of determining a strut for a given load rather than of finding the crippling load for a given strut, and accordingly proceed by the reverse of the method just suggested, starting with a curve of $\frac{d^2 y}{dx^2}$ assumed perfectly uniform as a first trial shape for

After stress varies with $\frac{d^2 y}{dx^2}$ and we wish to obtain uniform fiber stress, the assumed curve is shown in Fig. 2. When the curve is integrated with the integrator, then results a curve of $\frac{dy}{dx}$ and the integral of this second curve is a curve of dy itself, i.e., the elastic curve of the strut for a given load, and with the assumed $\frac{d^2 y}{dx^2}$ curve.

Equation (1) may be transformed to the following:

$$I = \frac{W y}{E \frac{d^2 y}{dx^2}} \text{ or } I = \frac{W y}{E k} \text{ where } k = \frac{W}{E}$$

In the preceding paragraph, k is assumed as 1 and the equation becomes:

$$I = \frac{W y}{E k}$$

and by means of this last equation, we obtain the moment of inertia of the several sections of the strut by dividing the ordinates of the deflection curve by those of the $\frac{d^2 y}{dx^2}$ curve, with due regard for instrument constants and the scales of the graphs. Thus from the moments of inertia, the diameters of the sections are obtained.

Lastly, from equation (2), it is seen that the fiber stress is proportional to the product of b and $\frac{d^2 y}{dx^2}$. Therefore, if for each section we multiply the diameter and value of $\frac{d^2 y}{dx^2}$ together, the product is proportional to the fiber stress at that section, and from the several products, a curve of relative fiber stress may be plotted.

Two times above procedure was carried out for the curve of Fig. 2, the fiber stress curve shown for fiber stress stress.

By successive trials, a curve of $\frac{d^2 y}{dx^2}$ was found, yielding a stress curve as shown in Fig. 3. This curve was considered satisfactory as a basis for the construction of the 1-strut.

It should be noted that the foregoing is based on the assumption that $X = \frac{W}{E} = 2$. From equation (1), we may obtain

$$- \frac{d^2 y}{dx^2} = \frac{W y}{E I}$$

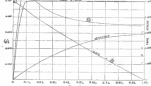


Fig. 3

and from that it is evident that when the ratio of W/E is changed, f must be changed in the same ratio, if the relation of x to $\frac{d^2 y}{dx^2}$ is not to be affected. Therefore, with a strut stress determined for a given load, it is a simple matter to design one for any other load, since it is only necessary to change the W of all the sections accordingly, with corresponding change in diameter.

MAKING THE STRUTS

A length of 60 inches was decided on for the struts, as this was about the maximum that could be handled easily in a lathe. The material was well seasoned, strength graded, western spruce. The struts were first planed straight, then turned down to within 0.04 inch of the correct diameters, using special cutting tools and steady rests designed for this purpose. The final handwork of an inch was removed by sandpaper.

The diameters to which the struts were made follow:

Section	From 0 to 10 in.	From 10 to 20 in.	From 20 to 30 in.	From 30 to 40 in.	From 40 to 50 in.	From 50 to 60 in.
Type A	1.000	1.000	1.000	1.000	1.000	1.000
Type B	1.000	1.000	1.000	1.000	1.000	1.000
Type C	1.000	1.000	1.000	1.000	1.000	1.000

Five cylindrical struts were made, three with a diameter equal to the maximum diameter of type A, and three with a diameter equal to the maximum of type B. These tapered struts were made of each of the types A, B, and C.

Fittings were machined from oak tree for the ends of the struts. These are shown in Fig. 4, where there is also shown the method used to receive the end-fitting in the hole of the testing machine. After using the large cylindrical struts, it was found that the hole of the fitting would not exactly suit if allowed to bear directly upon the base of the machine instead of resting in the saddle. It was also found that the hole should have a radius at its base $\frac{1}{8}$ inch. Furthermore, in reference to A and B, Fig. 4 it will be seen that a fitting of this type gives the strut a bending moment at the end, after the strain is released.

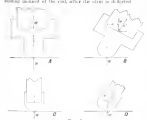


Fig. 4

The type of fitting was therefore discarded, and fittings of the type shown in Fig. 5 used thereafter. These fittings are so designed that the center of the tapered surface coincides with the center of the end of the strut, with the result that the line of action of the load W always passes through this point of the strut and any bending moment at the end and in the

middle. The failure of the second type of fitting is shown in Fig. 6 and in Fig. 7. These tapered fittings were used on the last six struts tested.

TESTING THE STRUTS

Both large and small cylindrical struts failed by crushing.

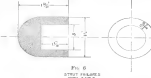


Fig. 5

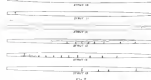


Fig. 6



Fig. 7

at mid-length. The large tapered struts of type A, as well as the small tapered struts of type B, failed by crushing near the ends, while the struts of type C failed by longitudinal shear and crushing at several points simultaneously. The failures of the struts of types B and C are shown in Fig. 7, while the loads and details of the tests are shown in the following table. The moments of inertia of the sections of type A are about three times as great as those of the same sections of type B, and it should be noted that the crippling loads of the former are about three times as great as those of the latter. All the tapered struts except those of type C were proportional from the curve shown in Fig. 2. In this figure it is seen that the fiber stress rises to a maximum between 0.10 and 0.20, which is where the strain broke. With this in mind, the last three struts (those of type C) were made slightly larger at the ends so that the stress dropped here, but remained practically constant throughout the remainder of the length. In these struts, failure occurred at several points at once. It is a significant fact that these failures always began along a sand-paper scratch.

CONCLUSIONS

A tapered strut cannot bear the same crippling load as a straight strut of the same maximum diameter and at the same time have a uniform fiber stress throughout its length. The tapered strut will stand only 80

* Abstract of a Thesis by C. F. Lewis and E. P. Haddock at the Massachusetts Institute of Technology, presented by Professor C. H. Fearing, June 1935. Reprinted by permission of the Massachusetts Institute of Technology.

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